DTU Library

Start-up performance of parabolic trough concentrating solar power plants

Ferruzza, Davide; Topel, Monika; Basaran, Ibrahim ; Laumert, Björn; Haglind, Fredrik

Published in:
AIP Conference Proceedings

Link to article, DOI:
10.1063/1.4984542

Publication date:
2017

Document Version
Publisher's PDF, also known as Version of record

Start-Up Performance of Parabolic Trough Concentrating Solar Power Plants

Davide Ferruzza[1,a)], Monika Topel[2,b)], Ibrahim Basaran[2)], Björn Laumert[2)], Fredrik Haglind[1)]

[1]*Department of Mechanical Engineering, DTU, Technical University of Denmark, 2800, Kgs. Lyngby, Denmark*
[2]*Department of Energy Technology, KTH Royal Institute of Technology, 100 44, Stockholm, Sweden*

[a)]Corresponding author: daferr@mek.dtu.dk
[b)]monika.topel@energy.kth.se

Abstract. Concentrating solar power plants, even though they can be integrated with thermal energy storage, are still subjected to cyclic start-up and shut-downs. As a consequence, in order to maximize their profitability and performance, the flexibility with respect to transient operations is essential. In this regard, two of the key components identified are the steam generation system and steam turbine. In general it is desirable to have fast ramp-up rates during the start-up of a power plant. However ramp-up rates are limited by, among other things, thermal stresses, which if high enough can compromise the life of the components. Moreover, from an operability perspective it might not be optimal to have designs for the highest heating rates, as there may be other components limiting the power plant start-up. Therefore, it is important to look at the interaction between the steam turbine and steam generator to determine the optimal ramp rates. This paper presents a methodology to account for thermal stresses limitations during the power plant start up, aiming at identifying which components limit the ramp rates. A detailed dynamic model of a parabolic trough power plant was developed and integrated with a control strategy to account for the start-up limitations of both the turbine and steam generator. The models have been introduced in an existing techno-economic tool developed by the authors (DYESOPT). The results indicated that for each application, an optimal heating rates range can be identified. For the specific case presented in the paper, an optimal range of 7-10 K/min of evaporator heating rate can result in a 1.7-2.1% increase in electricity production compared to a slower component (4 K/min).

INTRODUCTION

Concentrated solar power plants (CSPP) are foreseen to increase their share in the electricity production due to their ability to decouple the energy generation from the solar input. Although the majority of these plants are integrated with thermal energy storage (TES), they are still subject to daily start-ups and shut-downs [1]. Hence, in order to maximise the profitability of CSPPs it is essential that they are flexible with respect to transient operation. In this context, the most important components to study are the receiver, the turbine and the steam generator. The two former have been given attention in the research and studies on their dynamic performances have been conducted previously [2] [3], while the latter has not been the focus on any previous study, even though it represents a critical component, by being the connection link between the power block (PB) and the solar field (SF).

In current CSPPs, the steam generators have been designed as typical heat exchangers, and not as boilers suited specifically for CSP applications, requiring fast response times and high temperature ramp rates. Steam generators in CSP applications were firstly designed without focusing on the dynamic performance [4], as the industry mainly applied design of the components suited for conventional power generation practices. This has resulted in plants with start-up delays and a poor capacity to handle sudden changes in incident solar radiation or load demand, which in turn may cause failures in the steam generator due to excessive thermal stresses and deteriorate the economic viability of the plant. Currently, there is a tendency towards the development of steam generators tailored for CSPPs,

SolarPACES 2016
AIP Conf. Proc. 1850, 160008-1–160008-9; doi: 10.1063/1.4984542
Published by AIP Publishing. 978-0-7354-1522-5/$30.00

though there is still a lack of knowledge regarding the optimal ramp-up rate requirements. From the annual performance and competitiveness perspectives of a CSPP, it is desirable that both the turbine and the steam generator are able to start quickly, enabling the power plant to harvest the sun energy as soon as it becomes available. However, there may be limiting factors for the ramp-up time for one component, making it unnecessary to have a shorter ramp-up time for another. If, for example, the start-up rate of the turbine is the limiting factor, there is no need to design a boiler with a faster start-up rate than that of the turbine.

The objective of the paper is to identify which are the components that limit the start-up times for CSPPs during various operating conditions and evaluate their impact on the annual performance of the plant. In particular, the paper focuses on Parabolic Trough Power Plants (PTPP). Recent studies [5, 6], have focused their attention on the single component (i.e. turbine or steam generator), while in the work presented, both start-ups are considered, and their impact on the annual electricity production is evaluated. Firstly, the paper presents a brief description of the start-up limitations of the components. Secondly, in the methods section, the modelling of the PTPP and the operating strategy including the start-up are explained. Lastly, in the results section, the start-up performance of turbine and steam generator is analyzed together with the impact on the electricity production of the PTPP.

Nomenclature

Abbreviations

CSPP	Concentrated Solar Power Plant	**ST**	Steam Turbine
CT	Cold tank	**TES**	Thermal Energy Storage
D	Dearator	**WCC**	Wet cooled condenser
ECO	Economizer		
EVA	Evaporator		
HP	High pressure		
HT	Hot tank		
HTF	Heat Transfer Fluid		
HX	Heat Exchanger		
LP	Low pressure		
OPEX	Operation Expenditure		
PB	Power block		
PTPP	Parabolic Trough Power Plant		
REF	Reference case		
RH	Re-heater		
SF	Solar field		
SGS	Steam Generator System		
SH	Super-heater		

Subscripts

f	Fluid
Max	Maximum
Min	Minimum
nom	Nominal

Symbols	**Units**	
ITD	[°C]	Inlet Temperature Difference
m	[kg/s]	
p	[bar]	Pressure
T	[°C]	Temperature
t	[s]	Time
$\mathbf{v_T}$	[k/min]	Allowable ramp-up rate/heating rate

Steam Turbine and Steam Generator Start-up Description

The speed, at which both the steam generator system (SGS) and the steam turbine (ST) can start, is limited by constraints related to thermal stresses and low cycle fatigue. In both cases, these are related to thickness of components, material properties and temperature gradients [5] [6]. In the case of a steam turbine, the shaft seal and blading clearances define the allowable thermal expansion of the components. Typically, the starting up of a steam turbine can be divided in three phases: pre-start heating, running up and loading up. During the start-up the key parameter that limits the running and loading up speed is the difference in temperature between the incoming steam and turbine metal. Therefore, it is beneficial to keep these two temperatures as close as possible to avoid elevated thermal stresses in the component. As a consequence, the warmer the turbine material is before the start-up, the faster the start-up can be [5]. Manufacturers provide turbine loading curves governed by the metal temperature, to keep the stress within the allowable limits. Depending on the initial temperature of the turbine, (or on the turbine stand still time), the start-ups can be classified as hot, warm or cold. A hot start-up would take 8-10% of time of a full cold start-up while a warm start-up would range up to 45-50% [2].

In the case of steam generators, the main limiting factor for fast boiler start-ups are the maximum allowable stresses in the thick walled components such as headers of super-heaters, evaporator drum and T or Y junctions in the steam pipelines. The governing limiting component is the evaporator steam drum, as typically, it is designed as a high diameter vessel to withstand high pressures, and therefore large material thicknesses are required [6]. The start-up procedure in this case, is to reach the set points for steam temperature, pressure and mass flow rates as fast and as efficient as possible. Previous studies have pointed out that the main limiting components during the start-up, are either the evaporator or the super-heater [6] [7].Therefore these two will be considered in detail for the study.

METHODS

The study was carried out using DYESOPT, an in-house numerical tool developed at KTH, Royal Institute of Technology, Sweden, [2], allowing location-tailored, techno-economic performance evaluations integrated with multi-objective optimization of power plants. An annual performance analysis was carried out. As a first step, the start-up performance of the SGS and turbine was analyzed. The allowable start-up rates were included in the software by considering the allowable stresses on the most sensible parts of the steam generator (i.e. the steam drum and headers) [7] [8]. The allowable temperature gradients obtained for the thick-walled metal parts were used to determine the appropriate fluid temperature and pressure start up conditions [9]. Metal temperatures and start-up schedules provided by the manufacturer for the turbine were considered [2]. Secondly, the parabolic trough power plant (PTPP) was modeled by looking at its steady state design and dynamic performance. Lastly, the annual performance of such plant was analyzed, focusing on the start-up of the two components. This was considered by introducing in the model a control strategy, which accounted for the ramp-rates limitations. Based on this analysis, the optimal range of start-up rates of the components was determined.

Start-up Limitations for Turbine and Steam Generator

The start-up limitations of both components were considered by looking at the turbine loading curves for the different start-ups and by calculating the allowable fluid temperature change in the SGS components (EVA and SH).

Figure 1 illustrates the different start-up curves, in particular the A-B and B-C lines represents respectively the running up and loading up of the turbine [10]. The study analyzed different turbine ramp-up speeds, as well as the possibility to always ramp with a hot start-up. The latter requires that temperature maintaining modifications like electrically powered heat blankets or high speed barring are employed [5].

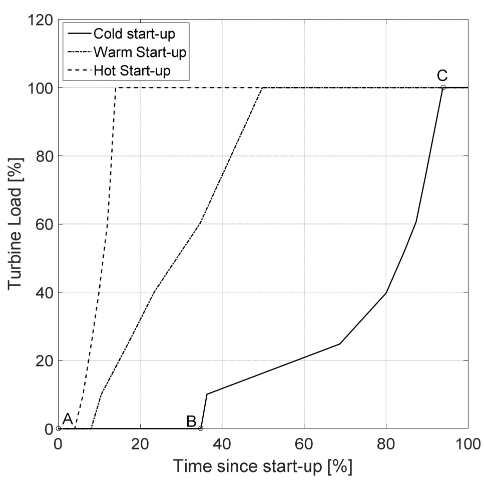

FIGURE 1: Turbine loading curves versus the percentage in time since start-up procedure has begun [10]

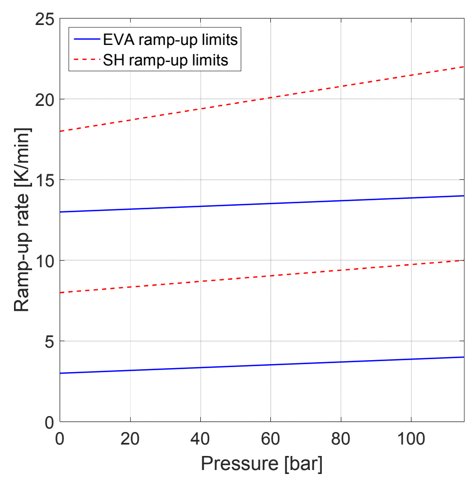

FIGURE 2: Assumed evaporator and super-heater ramp-up limits considered – For both components, the upper limit refers to a very fast case, while the lower limit refers to a slow case

Figure 2 presents different heating rates for both the super-heater and the evaporator. In both cases the upper and lower curves represent respectively the highest and lowest ramp-up rates that will be used in the yearly simulations of the power plant. The heating rates v_{Tmin} and v_{Tmax} respectively for the minimum and maximum allowable pressures can be determined according to the norm DIN EN 12952-3 [8]. In the case of this paper, the geometry of the components was not considered, instead an optimal ramp up rate from a system perspective was determined. The heating rates are used to calculate the allowable fluid temperature change using the following two equations [6]:

$$\frac{dT_f}{dt} = \frac{p_{max}\, v_{T_{min}} - p_{\,min}\;\; v_{T_{max}}}{p_{max} - p_{min}} + \frac{v_{T_{max}} - v_{T_{min}}}{p_{max} - p_{min}}\, p(T_f) \tag{1}$$

$$\frac{dT_f}{dt} = \frac{p_{max}\, v_{T_{min}} - p_{\,min}\;\; v_{T_{max}}}{p_{max} - p_{min}} + \frac{v_{T_{max}} - v_{T_{min}}}{p_{max} - p_{min}}\, p(t) \tag{2}$$

In the case of the evaporator, the water is at saturation point and therefore the pressure and temperature will be related. As a consequence, the temperature of the fluid will be dependent on the pressure and equation (2) can be solved with a Runge-Kutta method, assuming $T_f(t = 0\,) = T_0$ [6]. In the case of SH the fluid is not at saturation conditions and the pressure is a function of time determined by the evaporator conditions.

Parabolic Trough Power Plant Modeling and Design

The analysis of the annual performance and of the impact of the start-up performance was carried for PTPP. The layout of the power plant analyzed for the paper is shown in **Fig. 3.** The PTPP considered is integrated with an indirect TES system and a Wet Cooled Condenser (WCC). The power plant has been designed for the location of Seville, with a power output of 55 MWe gross. The design has been carried following ref. [11] for PB, ref. [12] for the HTF cycle and ref. [13] and ref. [14] for SF. The dynamic modeling was implemented in TRNSYS, as DYESOPT allows the coupling of steady state design in Matlab and the aforementioned software [5, 11].

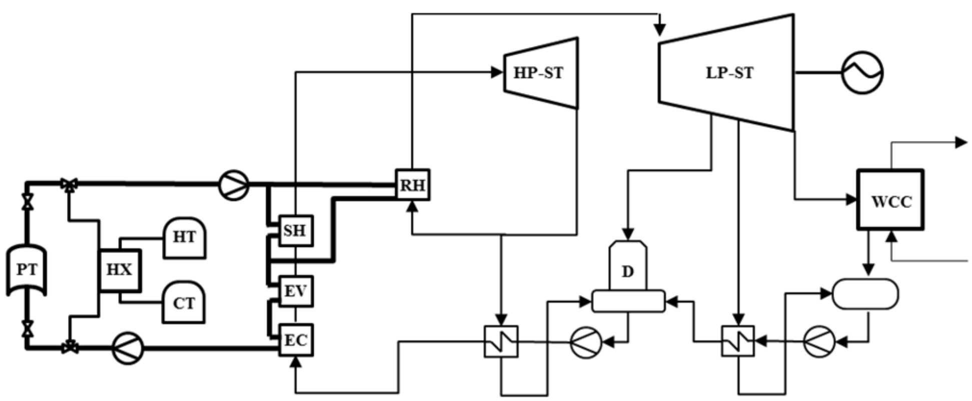

FIGURE 3: Layout of the considered parabolic trough power plant integrated with thermal energy storage and wet cooled condenser [13]

The main design parameters and thermal performance indicators are listed in **Table 1** and **Table 2**. **Table 2** also presents a comparison of the main performance indicators obtained with a similar PTPP model (SAM) [15].

The comparison with the data of the reference model [15] indicate that the largest deviation occurs for the yearly electricity production, and in this case the model predicts a value 8.9% lower than that of the reference model. However when considering comparison with ref. [16] and ref. [17] , the deviation ranges between +0.5% and -3.7%. Moreover in the reference model, the PTPP is integrated with an auxiliary burner to improve the production of electricity during start-ups or sudden losses in available thermal power [15] , which was not implemented in the model developed by the authors. These comparisons suggest that the models give results with sufficient accuracy for the purpose of the current paper. The power plant considered, will serve as a basis for further analysis of the start-up performance.

TABLE 1. Main design parameters for the analyzed PTPP

Main Design Parameters	Units	Value
SM	[-]	1.75
Gross Power Output	[MW_e]	55
TES Capacity	[h]	7.5
Inlet HP/LP-ST pressure	[bar]	100/16.5
SF Maximum outlet temperature	[°C]	393.3
Nominal WCC ITD	[°C]	11
# of HP/LP ST extractions	[-]	2/3
Operating strategy	[-]	Baseload

TABLE 2. Main performance indicators for the validation of the PTPP model

Performance indicators	Model results	Reference	Relative Error
Net Power Output [MWe]	49.97	49.90 [17]	+0.14%
PTPP annual average efficiency [%]	15.27	15.00 [17]	+1.8 %
Yearly electricity production [GWh]	158.9	158/165/174.5 [16] [17] [15]	+0.5%/-3.7%/ -8.9%
SF land area [hectares]	202.2	200 [17]	+1.1%

Control Strategy for Optimal Start-up

In order to implement their start-up performance, the control strategy, presented in **Fig. 4,** was implemented in the dynamic model of the power plant in TRNSYS. It comprises the strategy for turbine operation developed by the authors [7], and the operational strategy for steam generator in case of a heat transfer fluid (HTF).

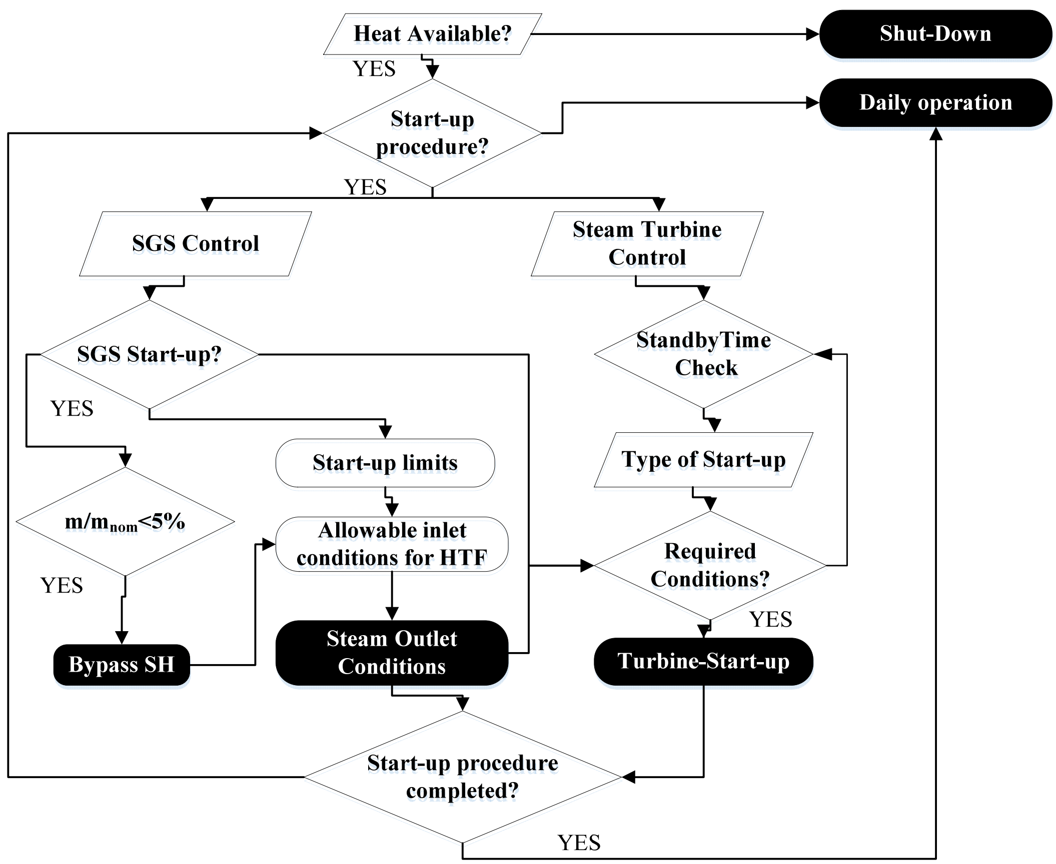

FIGURE 4: Control strategy logic diagram for start-up considering the interactions between steam generator and turbine

When the HTF is supplied to the evaporator, the system starts to produce steam, however the valves at the outlet of the drum are kept close until a certain amount of mass flow rate can be produced (5% of the nominal mass flow rate). This is to avoid overheating of the SH tubes. When the steam reaches acceptable conditions for the turbine (minimum pressure and temperature or degree of superheat), the control valves are opened and the turbine starts to run up, while pressure and temperature of the steam are increasing. Each type of turbine start-up has different acceptable conditions. For instance, a hot start-up would require higher steam inlet temperature than a warm or cold case. This is related to the fact that a hot start-up would imply a higher turbine metal temperature, therefore the acceptable steam temperature is higher to keep the difference between the two temperatures as low as possible and minimize the thermal stresses. After each shut down, the steam drum is able to keep the lowest admissible pressure of the turbine; therefore, aside from losses (which in this case are considered negligible), each consequent start-up will not begin from ambient pressure, but from the value set by the characteristics of the turbine. This always happens, unless a maintenance occurs, requiring start-up from ambient pressure and temperature.

During each start-up (either at the beginning of the day or due to sudden heat unavailability), the controller checks for turbine stand time and water conditions in the steam drum. This determines the type of turbine start-up (cold, warm or hot) and the allowable heating rate for the pressure reached at the evaporator. This signal is translated into the allowable oil inlet conditions (enthalpy, temperature and mass flow rate), according to heat availability from the SF and TES and the temperature reached by the oil. The outlet conditions from the super-heater are checked to determine whether the turbine can accept the steam in order to proceed for the start-up. Once both components reach their nominal operating points, the start-up procedure is completed.

RESULTS

Figure 5 presents the result of a start-up of the two components for the different cases presented in **Table 3**. The different cases are chosen to understand under which conditions, a start-up might be delayed. These are considered not only by looking at different SGS configuration (Cases a, b, c, d) but also at faster turbine (assuming a 20% faster than the reference case) (Cases 1, 2, 3, 4), to look if in any case the SH is not able to provide the nominal conditions when the turbine reaches full load. The figures illustrate temperature (for SH and EVA) and pressure (for EVA) evolution in time (represented in percentage in respect to the total simulation time) together with the turbine load.

TABLE 3. Summary of different start-up cases considering evaporator, super-heater and steam turbine ramp rates

	SH ramp-rates	EVA ramp-rates	Turbine ramping mode
Case a	Fast	Fast	For each case:
Case b	Slow	Fast	1) Warm start-up
Case c	Fast	Slow	2) 20% faster than the Warm start-up
Case d	Slow	Slow	3) Hot start-up
			4) 20% faster than the Hot start-up

The figures indicate that for cases a and b the SGS is able to provide the nominal conditions of steam when the turbine reaches full load, no matter what start-up mode is employed for the ST. However, it may be noted that a slow SH as in case b would postpone the start-up of the turbine as it is not able to provide sufficient degree of superheat to the outlet steam. On the other hand, for a too fast ramp-up of the SH at the beginning of the process as in case a, the SH would be constrained by the limitations of the EVA and consequently be ramped-up at a lower rate.

The comparison of case a and c suggests that a slower EVA could result in 2-18% lower start-up time, depending on the turbine conditions. However, cases b and d result in little differences (<0.5%), as the SH in case b is acting as the bottleneck of the entire start-up procedure, and reaching allowable turbine inlet conditions at the same time as in case d. These considerations suggest that the SH and the EVA should be optimally ramped up together and a fast SH is generally advisable. In the particular case of the PTPP under study, an optimal SH heating rate 179% higher than the EVA is found. In this way both components would reach the nominal point at the same time. Even if the design of the SH would allow higher ramp rates, operating at a slower rate would determine a lower associated fatigue and a longer durability of the component.

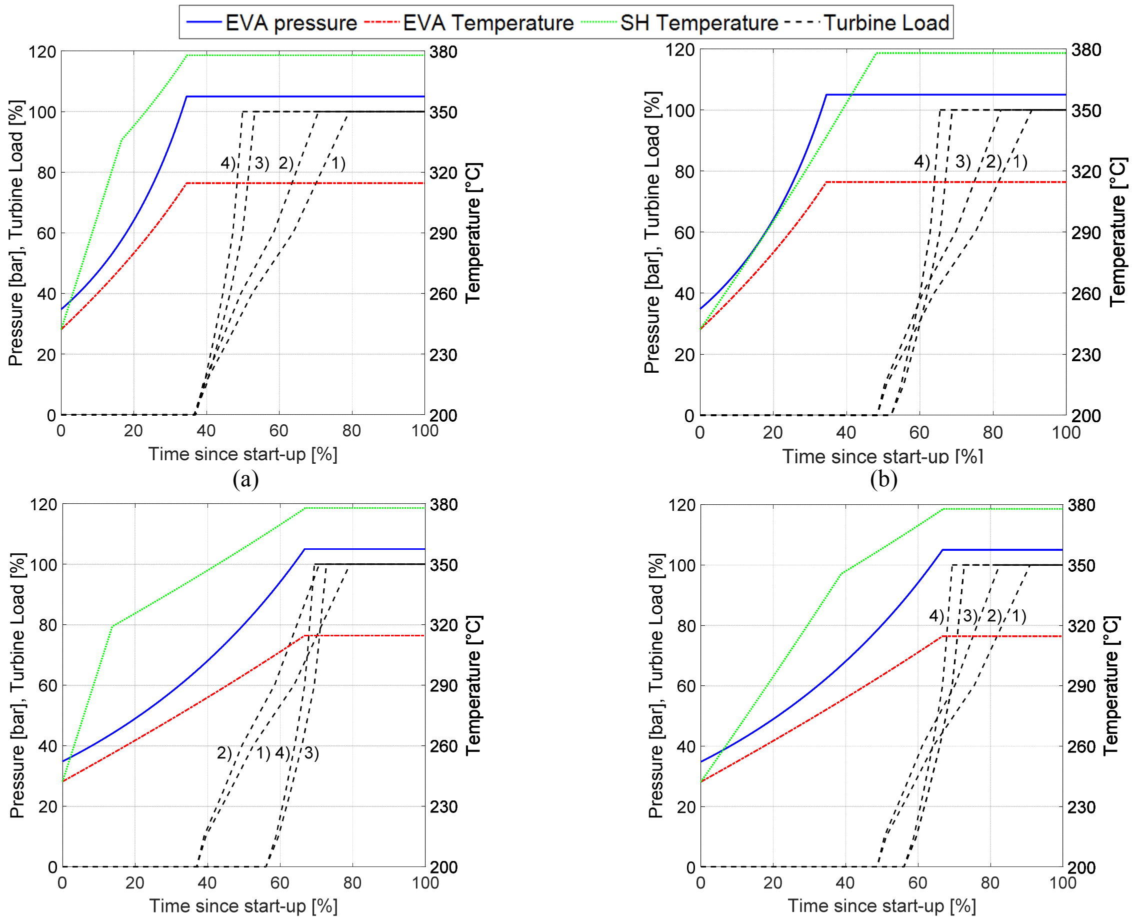

FIGURE 5. Start-up performance of evaporator, super-heater and turbine for the different cases – a) Case a – b) Case b – c) Case c – d) Case d

In order to assess the impact of the different ramp-rates on the yearly energy production of the PTPP, three different cases were compared with a reference case. The variation for EVA and SH ramp limits is illustrated in **Fig. 2**, while the turbine cases refer to **Table 3**. **Figure 6** presents the result of the analysis. Each point in **Fig. 6** represents the results of the annual dynamic simulation, deriving from different combinations of ramp rate design of the three components considered. The upper and lower curves represent the limits due to the different cases considered in the study, while the gray area represents all possible combinations in between. The key parameter chosen to be presented was the evaporator ramp-rate as it is the one with the higher influence in the results

Looking at the lower limit of the graph, a decreasing trend against the EVA ramp rates may be observed. This is due to the fact that the points in the bottom part of the graph are associated with low SH ramp rates values, making the SH the bottleneck of the start-up and postponing the turbine start-up procedure (as presented in Case b). On the contrary, the upper limit is characterized by an increasing trend, because each case is associated with the relative optimal SH ramp rate. Different cases are illustrated in **Fig. 6** and summarized in **Table 4**, to show the impact of the ramp rate conditions on the electricity production.

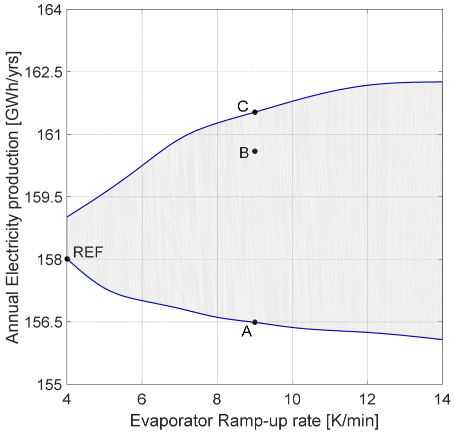

FIGURE 6: Annual electricity production considering different ramp-up rates for super-heater, evaporator and turbine

TABLE 4: Comparison of different cases for PTPP yearly performance

Case	Description	Electricity Production [GWh/yr]	Relative difference
(REF)	Slow EVA, Optimal SH, No turbine ramp modifications	158.01	0 %
(A)	Fast EVA, Slow SH, No turbine ramp modifications	156.49	-0.96 %
(B)	Fast EVA, Optimal SH, No turbine ramp modifications	160.6	+1.64 %
(C)	Fast EVA, Optimal SH, Only turbine hot start-ups	161.53	+2.23%

Each case (A, B, C) represents a possible combination of designs with a fast evaporator and compared to a slow reference case (REF), to show how a faster SGS could improve the performance of the power plant. Case A represents a non-optimal configuration shown as the bottom line of **Fig. 6**, suggesting that if the EVA and SH are not operated optimally together (or not designed properly), it would result in a nearly 1% loss in electricity production. If a fast optimally designed SGS (compared to the base case) is employed, the PTPP performance would benefit 1.64% in electricity production with a 0.59%-points potential increase if the turbine start-up improvements are considered. Note that having only hot turbine start-ups does not have a significant impact. This can be explained by looking at **Fig. 5a-b**. Even if the hot start-up is much faster, the steam temperature requirement is higher than in a warm case, postponing the actual start-up of the turbine and improving in a lower percentage the performance of the power plant than if the start-up could have occurred at the same instant as that of a warm start-up. Moreover from **Fig. 6**, it can be observed that an optimal range of start-up time for the evaporator can be identified. Indeed between 7-10 K/min the slope of the curve decreases to reach an asymptote for higher ramp-rates. This means that after this threshold, the increase in the heating rate of the component is affecting less the performance of the power plant. If, for example, a 14 K/min evaporator could be designed, it would result in a 0.46% increase (compared to case C) in electricity production, making it less worthwhile to reach for such limits.

CONCLUSIONS

A detailed methodology has been presented to show the interaction between steam turbine and steam generation system during the start-up of a PTPP. For this purpose a detailed model of a PTPP has been developed and implemented with a control strategy to account for the start-up limitations and performance of the aforementioned components. This has been validated and integrated in an existing tool for the dynamic performance evaluation of power plants (DYESOPT). The results suggested, that an optimal design and/or operating strategy accounting for both SH and EVA is crucial not to make the SGS the bottleneck of the power plant start-up. As such, for the particular application, it was found that a 179% higher start-up rate for the SH than the EVA is an optimal design/control point in order to both perform optimally the start-up procedure while affecting to a small extent the lifetime of the components. From a system perspective an optimal range of average heating rate for the evaporator has been found to lie between 7-10 K/min. With this configuration and an optimally operated super-heater, the increase in the yearly power plant energy production would result in 1.7-2.1%. However, an economic analysis comprising the OPEX related to the different SGS designs would be required to study the impact of such designs on the techno-economic indicators and profitability of PTPPs.

REFERENCES

1. CSP Today (Groupe Reaction Inc.), "CSP Solar Tower Report 2014: Cost, Performance and Thermal Storage," CSP Today, 2014.
2. M. Topel, M. Genrup, M. Jöcker and B. Laumert, "Operational improvements for startup time reduction in solar steam turbines," *Journal of Engineering for Gas Turbines and Power,* vol. 137, no. No. 4, 2015.
3. J. Samanes and J. Garcia-Barberena, "A model for the transient performance simulation of solar cavity receivers," *Solar Energy,* pp. 789-806, 2014.
4. L. L. Vant-Hull, "Central tower concentrating solar power (CSP) systems," in *Concentrating Solar Power Technology - Principles, Developments and Applications*, Woodhead Publishing Limited, 2012, pp. 240-283.
5. M. Topel, R. Guedez and B. Laumert, "Impact of increasing steam turbine flexibility on the annual performance of a direct steam generation tower power plant," in *SolarPaces*, 2014.
6. J. Taler, B. Weglowski, D. Taler, T. Sobota, P. Dzierwa, M. Trojan, P. Madejski and M. Pilarczyk, "Determination of start-up curves for a boiler with natural circulation based on the analysis of stresses distribution in critical pressure components," *Energy,* vol. 92, pp. 153-159, 2015.
7. I. Basaran, A comprensive study of the imposed limitations on concentrating solar power plant start-up speeds, Stockholm: KTH, Master Thesis, 2015.
8. European Standards, "EN 12952-3, Water tube boilers and auxiliary installations. Part 3: Design and calculation for pressure parts of the boiler;," DIN, 2011.
9. P. Dzierwa and J. Taler, "Optimum heating of pressure vessels with holes," *Journal of pressure vessel technology,* vol. 137, no. 1, p. 8, 2014.
10. J. Spelling, M. Jöcker and A. Martin, "Annual performance improvement for solar steam turbines through the use of temperature-maintaining modifications," *Solar Energy,* vol. 86, no. 1, pp. 496-504, 2012.
11. R. Guedez, M. Topel, I. Conde, F. Ferragut, I. Callaba, J. Spelling, Z. Hassar, C. D. Perez-Segarra and B. Laumert, "A methodology for determining optimum solar tower plant configurations and operating strategies to maximize profits based on hourly electricity market prices and tariffs," *Journal of Solar Engineering,* vol. 138, pp. 021006-1-11, 2016.
12. P. Gilman, N. Blair, M. Mehos, C. Christensen and S. Janzou, "Solar advisor model user guide for version 2.0," NREL, 2008.
13. F. Lippke, "Simulation of the Part-Load Behavior of a 30 Mwe SEGS Plant," Sandia National Laboratories, 1995.
14. V. A. Dudley, "SEGS LS-2 Solar Collector. Test Results," Sandia National Laboratories, 1994.
15. NREL, "System Advisor Model Case Study: Andasol-1," NREL, 2011.
16. NREL, "Concentrating Solar Power Projects - Andasol," 2013. [Online]. Available: http://www.nrel.gov/csp/solarpaces/project_detail.cfm/projectID=4. [Accessed 19 07 2016].
17. Solar Millenium AG, "The parabolic torugh power plants Andasol 1 to 3," Solar Millenium AG, 2008.
18. MAN Diesel & Turbo, "Solar Power Generation - Industrial steam turbines for CSP plants," MAN, 2011.